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TECHNICAL NOTE

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AN EMPIRICAL RELATION FOR PREDICTING VOID FRACTION WITH TWO-PHASE, STEAM-WATER FLOW

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SUMMARY

A review is made of some of the experimental data and recent analyses applicable to predicting void fractions with two-phase, steam-water flow through tubes and channels. An empirical equation is presented that closely approximates experimentally measured void fractions in terms of the fluid quality and the ratio of the density of the fluid phases. Correlation of data is achieved over a range of fluid pressures from 1 to over 135 atmospheres and fluid qualities from 0 to 1.0.

INTRODUCTION

In considering two-phase-flow pressure-drop problems it is necessary to know the void fraction of the flowing fluid. Void fraction in two-phase flow is generally defined as the volume ratio of vapor to total fluid, whereas the fluid quality is defined as the mass ratio of vapor to total fluid. The void fraction cannot be calculated directly from the relation of the ratio of vapor-to-liquid density and the fluid quality, as will be shown later herein, because of the unknown local ratio of vapor-to-liquid velocity (velocity slip ratio) in two-phase flow. In addition, effects of void fraction distribution across the tube cross section and possible hydrodynamic forces existing between the liquid and the vapor bubbles also complicate the direct calculation of void fraction from a determination of the thermodynamic quality.

In considering theoretical flow models for representing void fractions in two-phase flow, some investigators have assumed that the fluids are separated into two parallel conduits, one liquid and one vapor or bubbles. However, experiments have shown that the two phases of a fluid are neither completely interspersed nor completely separated into such conduits. Even where strong mixing of the phases occurs, radial density variations are observed. Other investigators have assumed that the two phases are sufficiently well mixed that they flow as a single homogeneous fluid, but that some radial gradients of density and other physical properties exist in a direction normal to the flow.

Because of the difficult nature of the problem of describing the homogeneity of the two-phase flow, an empirical rather than theoretical approach has been taken herein for determining void fraction.

A brief review is made of some of the published void fraction data for steam-water flow in tubes and channels, and pertinent theoretical analyses are indicated. An empirical equation is then presented from which void fraction values can be calculated that correlate closely with experimental data over a fluid pressure range from 1 to over 135 atmospheres and fluid qualities from 0 to 1.0.

BASIC MODEL

For a flowing system the average void fraction α for a unit tube length large enough to include a representative sample of the fluid in the tube can be written as:

$$\alpha = \frac{A_v}{A} = \frac{A_v}{A_v + A_L} = \frac{1}{\frac{A_L}{A_v} + 1} \qquad (1)$$

where A is the cross-sectional area for each phase and the subscripts v and L denote vapor and liquid, respectively. In terms of the weight flow w through a tube, the cross-sectional areas occupied by each phase at a specific station along the tube can be expressed by:

$$A_v = \frac{w_v}{\rho_v V_v} \quad \text{and} \quad A_L = \frac{w_L}{\rho_L V_L}; \; \beta = \frac{\rho_v}{\rho_L} \qquad (2)$$

where ρ is the phase density, V is the phase velocity, and β is the density ratio of vapor to liquid. The fluid quality x can then be expressed as:

$$x = \frac{w_v}{w} = \frac{w_v}{w_v + w_L} \qquad (3)$$

and

$$w_v = xw; \qquad w_L = (1 - x)w \qquad (4)$$

From the relations given by equations (1) to (4), the void fraction can then be written as:

$$\alpha = \frac{1}{\dfrac{(w_L/\rho_L V_L)}{(w_V/\rho_V V_V)} + 1} = \frac{1}{\dfrac{\left[\dfrac{(1 - x)w}{\rho_L V_L}\right]}{(xw/\rho_V V_V)} + 1} = \frac{1}{\left(\dfrac{1 - x}{x}\right)\beta\left(\dfrac{V_V}{V_L}\right) + 1} \tag{5}$$

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The ratio of V_V/V_L in equation (5) is generally called the two-phase velocity slip ratio. It is primarily this unknown velocity slip ratio, as well as effects of void distribution across the tube cross section and possible hydrodynamic forces existing between the liquid and the vapor bubbles and not included in equation (5), that prevents the direct calculation of void fraction from a determination of the thermodynamic quality.

For two-phase flow, in which the velocity slip ratio is substantially 1.0 (homogeneous mixture) and the effects of void distribution and hydrodynamic effects are neglected, the void fraction can be calculated simply by:

$$\alpha = \frac{1}{\left(\dfrac{1 - x}{x}\right)\beta + 1} \tag{6}$$

Equation (6) for steam-water calculations yields void fraction values higher by about 10 percent than those obtained experimentally in the literature (refs. 1 to 10) for fluid qualities greater than 0.1, indicative of the fact that at the higher qualities two-phase mixtures are substantially homogeneous and the velocity slip ratio V_V/V_L is actually near 1.0. For decreasing quality values less than 0.1 the differences between the void fractions calculated with equation (6) and those obtained experimentally become increasingly greater except, of course, near zero quality. It would appear then that the problem in the determination of the void fraction for two-phase pressure loss calculations is primarily confined to the low-quality regime.

A recent theoretical analysis concerned with the calculation of void fraction is presented in reference 1. Therein the two-phase flow model is assumed to consist of a dispersion of very small bubbles in a liquid. For such a model, steady-state transport conditions for these bubbles can be formulated, and, from these, complex expressions of the radial variation in bubble concentration and mean fluid velocity can be obtained. Since measurements of momentum and bubble diffusivities to substantiate such expressions are not available, reference 1 assumes a power-law representation across a circular tube cross section for the void fraction and the local velocity distribution in a single-phase turbulent fluid. These power-law representations are given by:

$$\left.\begin{aligned} u/u_m &= (y/R)^{1/m} \\ \alpha/\alpha_m &= (y/R)^{1/n} \end{aligned}\right\} \qquad (7)$$

where u_m and α_m are the velocity and void fractions at the center of a tube or channel whose radius is R; y is the distance from the tube wall; and m and n are empirical positive exponents.

From a consideration of liquid and vapor mass-flow rates and these power-law representations, an equation is presented in reference 1 that relates the average void fraction α to the mass fraction of the vapor phase or quality x, the ratio of the vapor-to-liquid density β, and a flow parameter K as follows:

$$\frac{1}{x} = 1 - \frac{1}{\beta}\left(1 - \frac{K}{\alpha}\right) \qquad (8)$$

The flow parameter K is a function of m and n with a range of variation from about 0.6 to 1.0 (ref. 1).

It is further suggested in reference 1 that the void fraction distribution exponent n should be a function of fluid pressure, since the tendency for a vapor to concentrate in the center of a tube or channel depends on the ratio of the densities of the fluid phases. Therefore, pressure or density should have some effect on the flow parameter K.

Data of sufficient detail and refinement are not available to permit an evaluation of the velocity and void fraction profiles in terms of the necessary variables. Consequently, an effort has been made to develop an empirical equation similar to that obtained theoretically in reference 1 (see eq. (8)) but including the concept that void fraction is a function of the ratio of the densities of the fluid phases. Experimental data useful for developing such an empirical equation are contained in references 2 to 7, as well as being summarized in reference 1. Based on these data and considerations, the following empirical equation was evolved:

$$\frac{1}{x} = 1 - \left(\frac{1}{\beta}\right)^{0.67}\left[1 - \left(\frac{1}{\alpha}\right)^{(1/\beta)^{0.1}}\right] \qquad (9)$$

This equation is similar to that given in reference 8 in that the ratio of the density of the fluid phases appears with an exponent of 2/3, but differs in that the void fraction appears with a variable exponent that is also a function of the ratio of the densities of the fluid phases.

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COMPARISON OF RESULTS FROM PRESENT WORK WITH EXPERIMENTAL DATA AND OTHER ANALYSES

A comparison of the calculated results obtained with the present empirical equation (9) and those obtained with the analyses presented in references 1, 9, and 10 together with the experimental data of references 2 to 7 are shown in figures 1 to 6. The results are shown as the variation of void fraction with fluid quality for various fluid pressure values. It is seen that the present work represents the experimental data reasonably well over the entire range of the available data including that for $x < 0.1$. For the most part the present work is in reasonable agreement with that of reference 1. However, the theoretical equations used in reference 1 yield consistently low void fraction values compared with experimental data for void fractions greater than 0.6, whereas the present work continues to represent void fractions obtained experimentally up to the limit of 1.0. The analysis presented in reference 9 yields void fractions considerably lower than those obtained experimentally, expecially for void fraction values less than 0.6. The analysis of reference 10 shows reasonable agreement with the experimental data for void fractions greater than 0.7; however, at lower void fraction values a pressure dependency is noted that results in calculated void fraction values either higher or lower than those obtained experimentally.

Some investigators, including reference 9, imply in their theories that a basic difference exists in the flow structure in horizontal and vertical flow. The limited experimental data available appear to show some evidence that this assumption is valid; however, the scatter of the applicable data and the gross techniques used in obtaining these data are such as to preclude any final conclusions at this time. The present work, as well as that of reference 1, does not differentiate between horizontal and vertical flow structures. It is apparent, however, from the data shown in figures 1 to 6 that the present work appears to be applicable, at least for operational and engineering purposes, to either flow structure as well as multiple channels of other than circular cross section (see fig. 2).

CONCLUDING REMARKS

The present work, as well as the theoretical studies referenced, does not account for possible effects of heating rate and flow rate on the variation of void fraction with fluid quality. A study of the experimental data in references 3, 4, and 7, in which heat addition was used in order to obtain two-phase flow, suggests a grouping of the data as a presently undetermined function of both the heating and the flow rates. In addition, fluids with properties much different from water-steam flow should be investigated to establish the significance of fluid properties on the variation of void fraction with fluid quality.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, October 27, 1961

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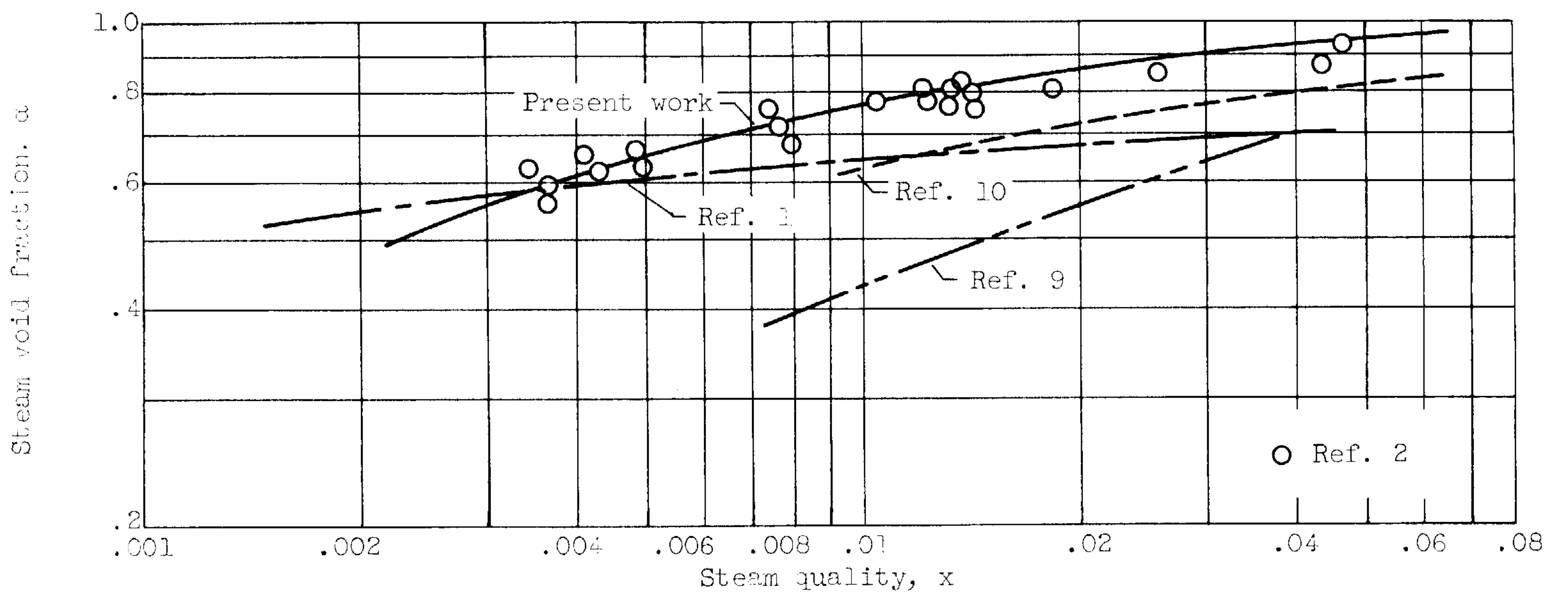

Figure 1. - Comparison of present work with data of reference 2 and with theories of references 1, 9, and 10. No heat addition; fluid pressure, 14.7 pounds per square inch absolute.

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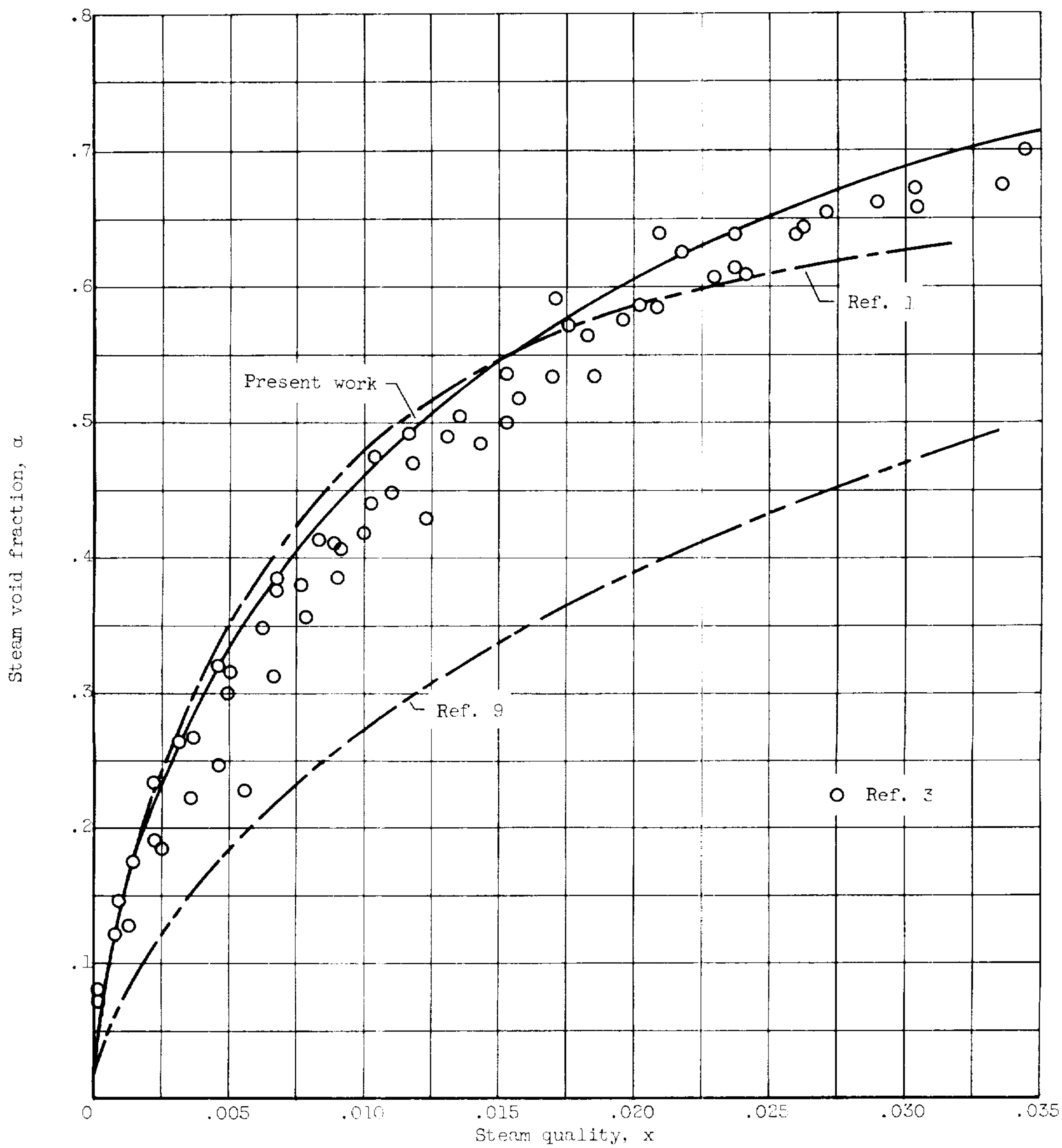

(a) Fluid pressure, 115 pounds per square inch absolute.

Figure 2. - Comparison of present work with data of reference 3 in multiple rectangular channels with varying heat addition and with theories of references 1 and 9.

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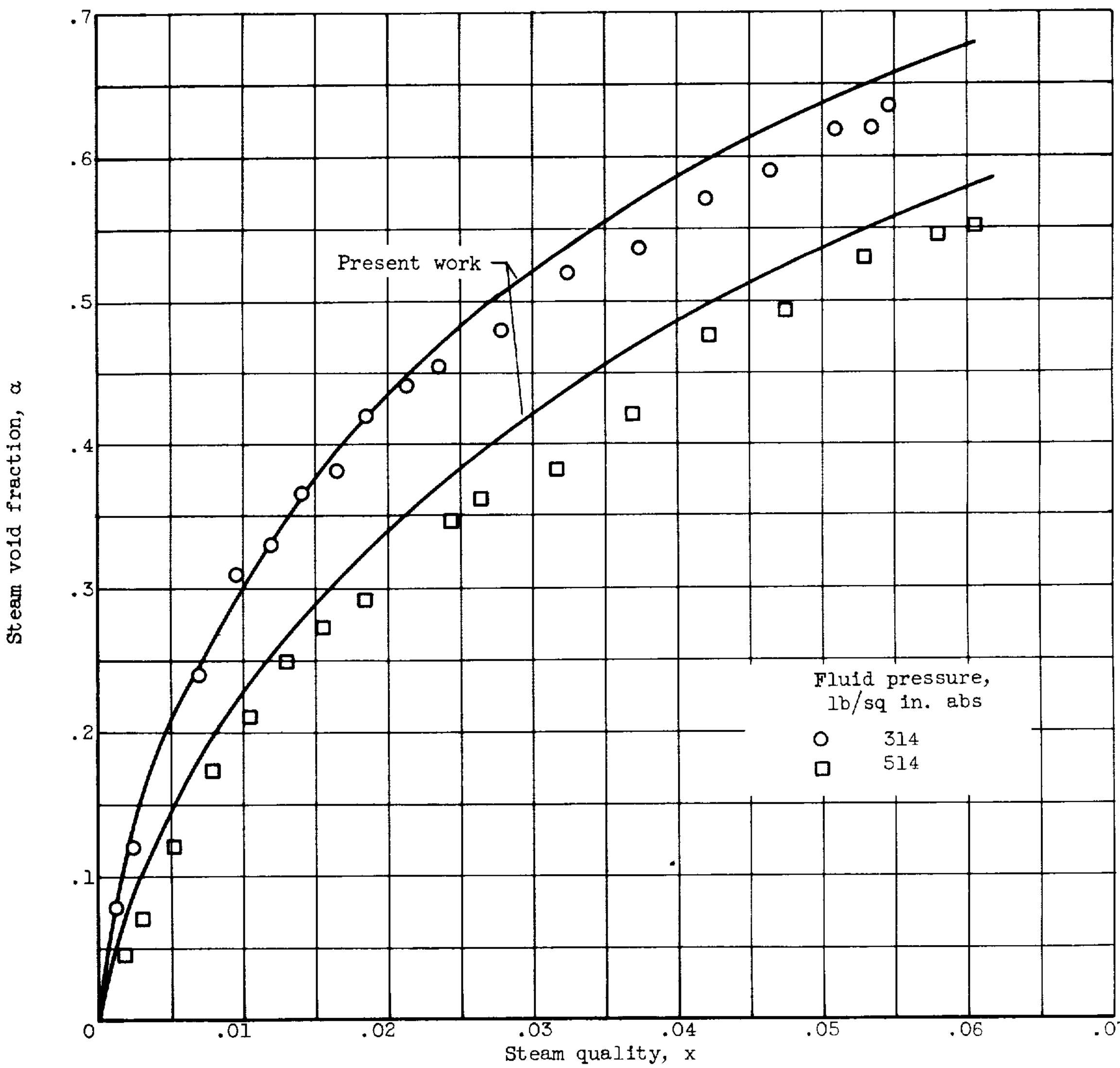

(b) Fluid pressure, 314 and 514 pounds per square inch absolute; reference 9.

Figure 2. - Concluded. Comparison of present work with data of reference 3 in multiple rectangular channels with varying heat addition and with theories of references 1 and 9.

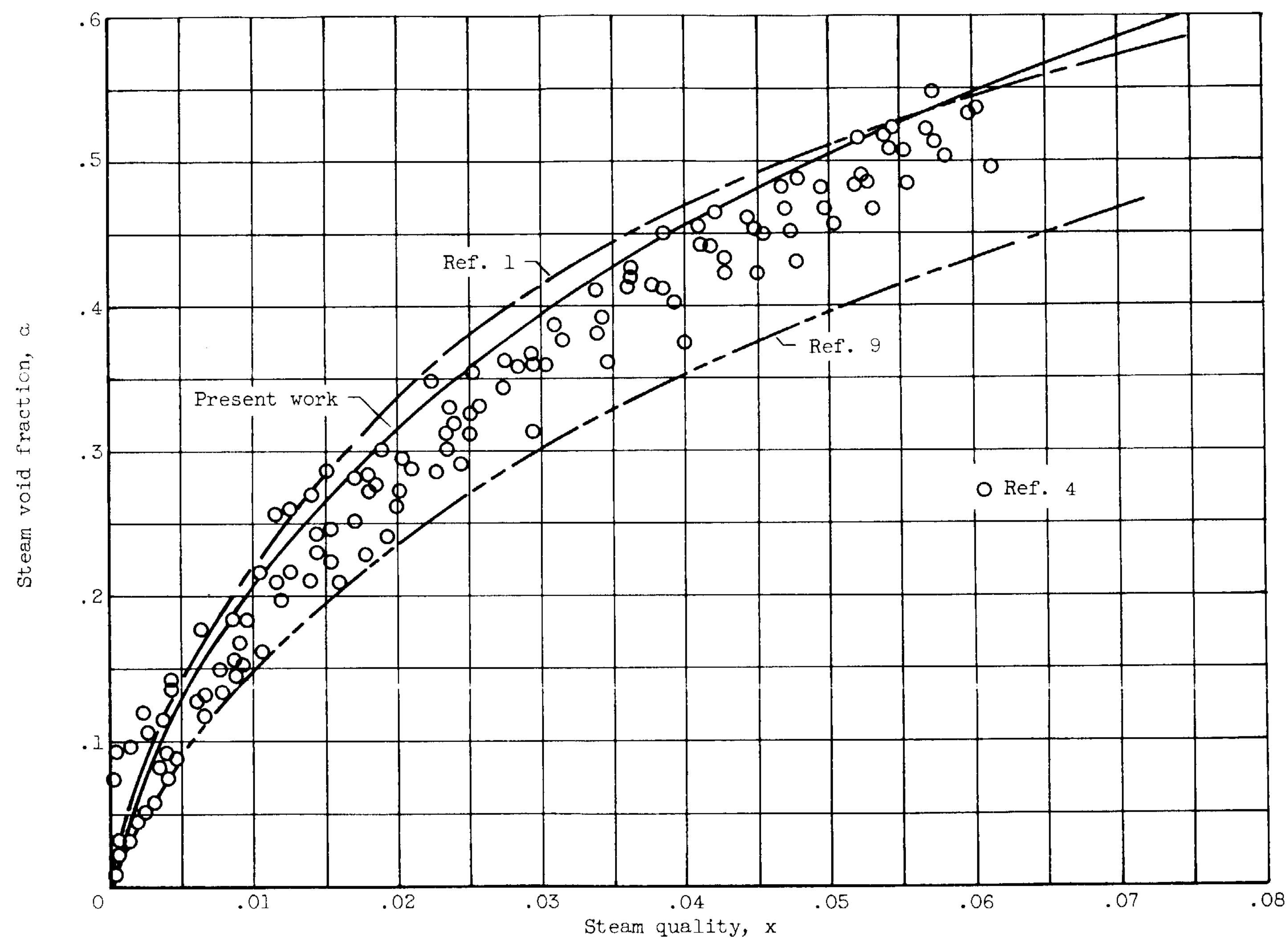

Figure 3. - Comparison of present work with data of reference 4 at 600 pounds per square inch absolute and theories of references 1 and 9. Multiple rectangular channels; vertical flow; varying heat addition.

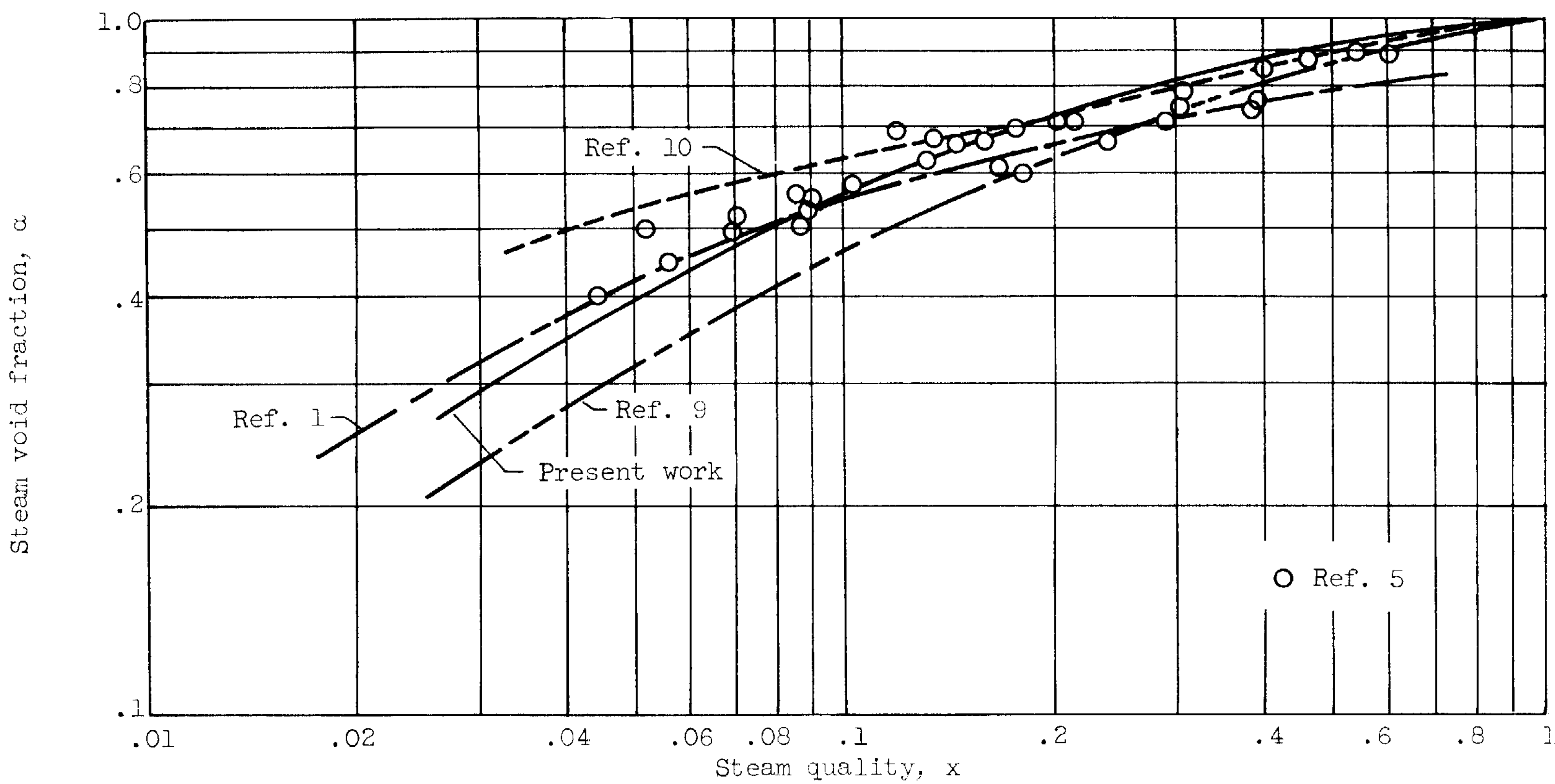

Figure 4. - Comparison of present work with data of reference 5 in horizontal steam-water flow at 1000 pounds per square inch absolute without heat addition and with theories of references 1, 9, and 10.

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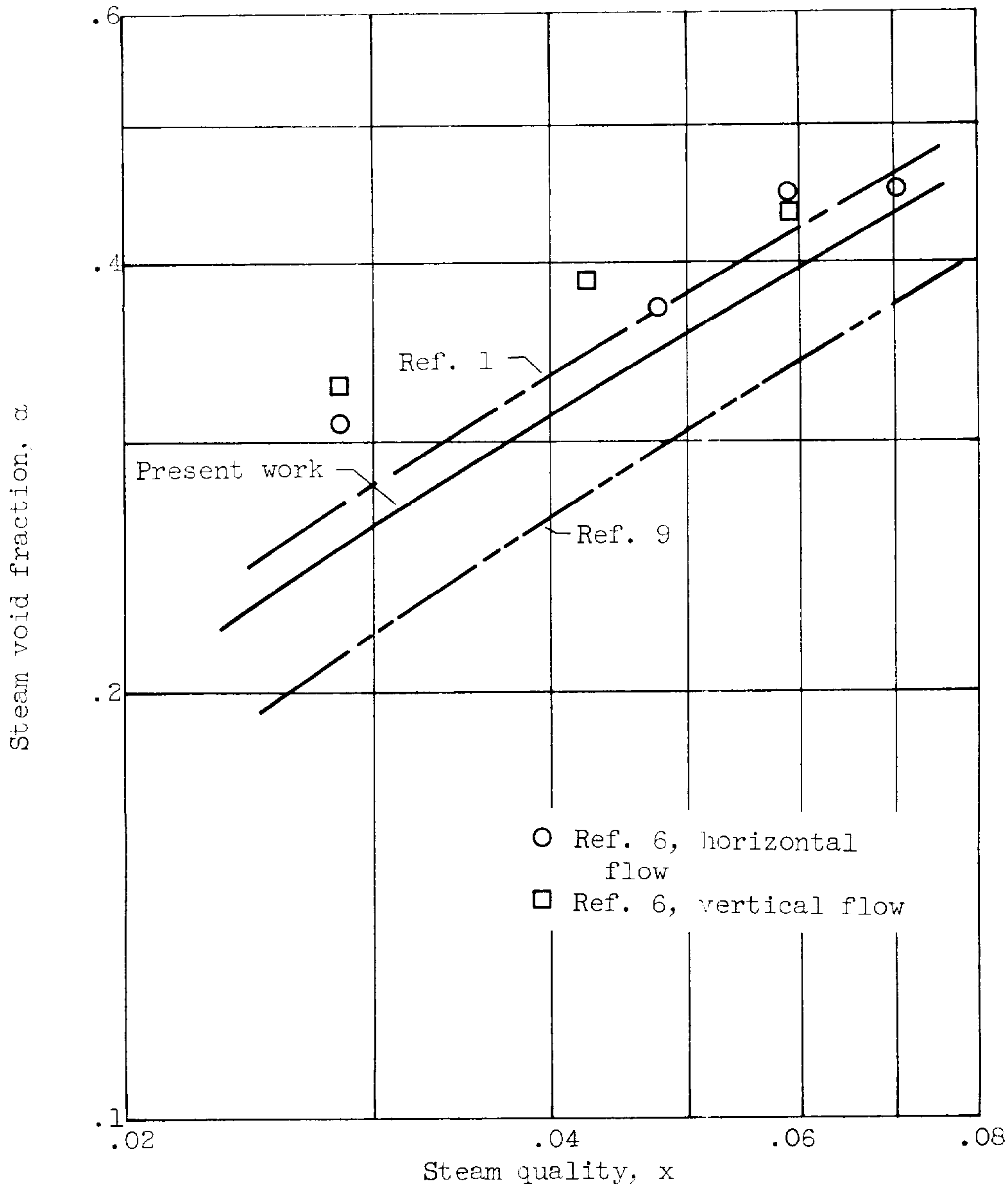

Figure 5. - Comparison of present work with data of reference 6 and with theories of references 1 and 9. No heat addition; fluid pressure, 1180 pounds per square inch absolute.

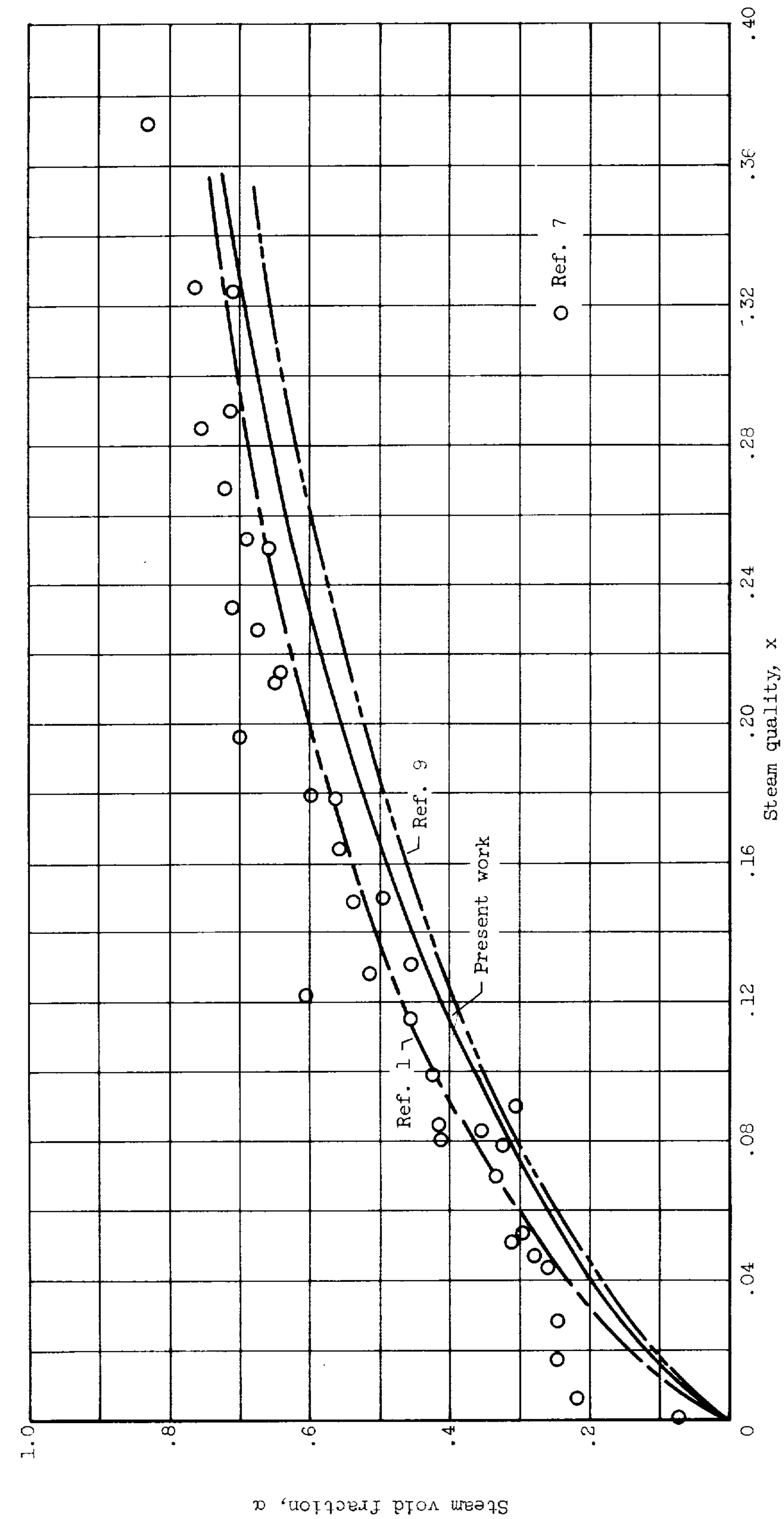

Figure 6. - Comparison of present work with data of reference 7 at 200 pounds per square inch absolute and with theories of references 1 and 9. Varying heat addition.

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